SUBSTITUTE SPECIFICATION

DESCRIPTION

HERMETIC COMPRESSOR

TECHNICAL FIELD

The present invention relates to a hermetic compressor used in a refrigerating cycle of a refrigerator freezer, etc.

BACKGROUND ART

Recently, reduction in power consumption of this kind of hermetic compressor has been strongly demanded. In a hermetic compressor disclosed in International Publication WO 02/02944, by improving an outer shape of a piston, sliding loss between a piston and a cylinder is reduced so as to achieve efficiency.

Hereinafter, a conventional hermetic compressor is described with reference to the drawings.

Fig. 7 is a longitudinal sectional view showing a general hermetic compressor described in US Patent No. 5,228,843; and Fig. 8 is a perspective view showing a piston described in International Publication WO 02/02944.

As shown in Fig. 7, hermetic housing 1 houses motor element 4 consisting of stator 2 having winding portion 2a and rotor 3, and compression element 5 driven by motor element 4. Moreover, in the lower part of hermetic housing 1, oil 6 is contained.

Crankshaft 10 includes main shaft 11 to which rotor 3 is press-fitted and fixed and eccentric shaft 12 formed eccentric to main shaft 11. Inside main shaft 11, oil pump 13 is housed and an opening portion of oil pump 13 is disposed in oil 6. Block 20 provided at the upper side of motor element 4 has cylinder 21 having a substantially cylindrical shape and bearing 22 for

supporting main shaft 11. Piston 30 is inserted into cylinder 21 of block 20 a capable of reciprocating sliding, and is coupled to eccentric shaft 12 via connecting means 41.

Next, a conventional piston is described with reference to Fig. 8. Piston 30 includes top surface 31, skirt surface 32 and outer circumferential surface 33. Furthermore, outer circumferential surface 33 includes seal surface 34, two guide surfaces 35 and removed portions 36. Herein, seal surface 34 is a surface in the circumferential direction, which is formed so as to be brought into close contact with the inner circumferential surface of cylinder 21. Guide surface 35 is formed so as to be brought into close contact with a part of the inner circumferential surface of cylinder 21 and extends substantially in parallel in the direction of the movement of piston 30. Removed portion 36 is a concave portion that is not brought into close contact with the inner circumferential surface of cylinder 21. Furthermore, an angle made by lines respectively connecting between central axis 37 of cylindrical piston 30 and two boundary edges 35a and 35b of guide surface 35 in the direction of the radius of piston 30 is generally 40° or less and preferably 30° or less.

Next, an operation of a conventional hermetic compressor shown in Fig. 7 is described.

During operation, piston 30 reciprocates in the horizontal direction in the drawing. In the vicinity of the bottom dead center, a part of the skirt side of piston 30 is protruded to the outside of cylinder 21. From this state, when piston 30 enters cylinder 21, that it is to say, when piston 30 moves in the right direction of Fig. 7, piston 30 is guided by guide surface 35 and thereby can enter cylinder 21 smoothly.

However, in a conventional hermetic compressor, inclination in the

vertical (perpendicular) direction of piston 30 with respect to cylinder 21 is regulated by space between outer circumferential surface 33 and cylinder 21 only in short section 34A between the edge of top surface 31 and the edge of seal surface 34. Therefore, piston 30 is likely to be inclined in the vertical direction. In particular, during the compression stroke from the bottom dead center to the top dead center (movement in the right direction in Fig. 7), top surface 31 of piston 30 undergoes compression load of a refrigerant gas and furthermore, crankshaft 10 is pressed in the direction that is not the direction of a piston (downward direction in Fig. 7) via connecting means 41, and thereby the inclination of piston 30 in the vertical direction is likely to be increased. As a result, there has been a problem that leakage of refrigerant increases, and the refrigerating capacity is deteriorated so as to lower the efficiency.

In particular, when low-density refrigerant Isobutane (R600a) was used, the outer diameter of piston 30 was increased and leakage of refrigerant was likely to occur, and therefore the efficiency was lowered remarkably.

SUMMARY OF THE INVENTION

In order to solve the above-mentioned problems of the prior art, a hermetic compressor of the present invention includes an under cut that does not communicate with at least a top surface of a piston on an outer circumferential surface of the piston excluding a sliding surface provided in the axis direction and in the perpendicular direction of the piston pin, in which the under cut communicates with space inside a housing at least in the vicinity of the bottom dead center. This configuration makes it possible to reduce sliding loss due to the reduction in a sliding area. Furthermore, by the sliding surface provided in the parallel and in the perpendicular direction of the piston pin, the inclination of the piston with respect to the cylinder is suppressed, thus

suppressing the leakage of refrigerant. Furthermore, by supplying the sliding portion with oil through the under cut, the sealing property can be improved. With the above-mentioned effect, a hermetic compressor with high efficiency can be provided.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 is a longitudinal sectional view showing a hermetic compressor in an exemplary embodiment of the present invention.

Fig. 2 is an enlarged sectional view showing an element around a piston used for a hermetic compressor in an exemplary embodiment.

Fig. 3 is a front view showing a piston used for a hermetic compressor in an exemplary embodiment.

Fig. 4 is a sectional view of a part along line 4-4 of FIG. 3.

Fig. 5 is an enlarged sectional view showing an end face of an under cut of a piston used for a hermetic compressor in an exemplary embodiment.

Fig. 6 is an enlarged sectional view showing a tip of a piston used for a hermetic compressor in an exemplary embodiment.

Fig. 7 is a longitudinal sectional view showing a conventional hermetic compressor.

Fig. 8 is a perspective view showing a piston used for a conventional hermetic compressor.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Hereinafter, an exemplary embodiment of the present invention is described with reference to the drawings. Note here that the present invention is not limited by the exemplary embodiment.

(EXEMPLARY EMBODIMENT)

Fig. 1 is a longitudinal sectional view showing a hermetic compressor in an exemplary embodiment of the present invention; Fig. 2 is an enlarged sectional view showing an element around a piston; Fig. 3 is a front view showing the piston; Fig. 4 is a sectional view of a part along line 4-4 of FIG. 3; Fig. 5 is an enlarged sectional view showing an end face of an under cut of the piston; and Fig. 6 is an enlarged sectional view showing a tip of the piston.

As shown in Figs. 1 to 6, housing 101 houses motor element 104 and compression mechanism 105 driven by motor element 104, and moreover contains oil 106. Motor element 104 includes stator 102 and rotor 103, and enables inverter driving by using a control circuit, etc. controlled at plural operational frequencies including operation frequency that is not higher than power supply frequency.

The hermetic compressor of this exemplary embodiment uses hydrocarbon-based refrigerant Isobutane (or R600a). Refrigerant R600a is a natural refrigerant with low global warming potential.

Crankshaft 110 includes main shaft 111 and eccentric shaft 112 and is disposed in a substantially vertical direction. Herein, rotor 103 is press-fitted and fixed to main shaft 111 and eccentric shaft 112 is disposed eccentric to main shaft 111.

Oil supplying structure 120 includes centrifugal pump 122, vertical hole 123, and lateral hole 124. One end of centrifugal pump 122 formed inside of crankshaft 110 is opened in oil 106 with another end connected to viscosity pump 121. One end of vertical hole 123 is connected to one end of viscosity pump 121 with another end opened in space inside housing 101.

Block 130 includes substantially cylindrical cylinder 131, main bearing 132 for supporting main shaft 111 and collision-portion 134 provided on the

upper side of cylinder 131. Cylinder 131 includes notch 135 provided on the upper side of the edge at the side of crankshaft 110.

Piston 140 is inserted into cylinder 131 and is capable of reciprocating sliding. Piston 140 has piston pin hole 141 formed in parallel to the center axis of eccentric shaft 112. Into piston pin hole 141, hollow cylindrical piston pin 142 is fitted. Piston pin 142 is fixed to piston 140 by hollow cylindrical lock pin 143. Piston pin 142 is connected to eccentric shaft 112 via connecting rod 146.

Hollow part 144 of piston pin 142 communicates with space inside housing 101 via vent hole 145.

On outer circumferential surface 150 of piston 140, under cuts 153 are formed. Each under cut 153 does not reach top surface 151 of piston 140 but reaches skirt surface 152. Fig. 4 is a sectional view of a part of piston 140 taken along line 4-4 of FIG. 3, showing a state of cylindrical central axis 170 of the piston seen from the left direction. As viewed in Fig. 4, under cut 153 is formed in the outer circumferential surface outside a region of a predetermined width and existing in parallel direction 147 with respect to the axis of piston pin 142 and outside a region of a predetermined width and existing in the perpendicular direction 148 with respect to the axis of piston pin 142. Total area of under cut 153 occupies not less than one half of an area of outer circumferential surface 150 of the piston. Furthermore, as shown in Fig. 5 that is an enlarged view showing the vicinity of edge 180 of under cut 153, angle θ made by edge 180 of under cut 153 and outer circumferential surface 150 of the piston is set to be an acute angle.

Furthermore, as shown in Fig. 3, the right end portion of piston 140 is provided with circumferentially formed land 190, on which under cut 153 is not formed, in a predetermined width from top surface 151. Furthermore, outer

circumferential surface 150 that does not belong to any of circumferentially formed land 190 and under cut 153 is referred to as axially formed land 192. In Fig. 3, axially formed land 192 is provided in parallel to cylindrical central axis 170 and extends from circumferentially formed land 190 and reaches skirt surface 152. As shown in Fig. 4, axially formed lands 192 are formed in a predetermined width on an outer circumferential surfaces at 0°, 90°, 180° and 270° with respect to the cylinder axis as a center.

Furthermore, as shown in Fig. 4, it is preferable that the width of axially formed land 192 is set so that angle ω made by two lines linking between cylindrical central axis 170 of piston 140 and two boundary portions of axially formed land 192 in the direction of radius of the piston is set to 40° or less and preferably 30° or less.

As shown in Fig. 4, in outer circumferential surface 150 of the piston, upper sliding surface 154 and lower sliding surface 155 are provided in the vertical direction and side sliding surface 160 is provided in the direction of the side surface. These correspond to one or both of circumferentially formed land 190 and axially formed land 192.

Furthermore, on circumferentially formed land 190, two annular grooves 191 are provided in the outer circumferential direction of the piston. Furthermore, on outer circumferential surface 150 of the piston, at both end portions of the top surface 151 side and the skirt surface 152 side, minute tapers 201 and 202 are provided.

In this exemplary embodiment, as shown in Fig. 1, in the vicinity of the bottom dead center, a part of the skirt side of piston 140 is protruded from cylinder 131. With such a configuration, even in a shape in which under cut 153 does not reach skirt surface 152, under cut 153 is opened in space inside the housing when at least piston 140 is in the bottom dead center.

Next, the operation and action of the hermetic compressor of the exemplary embodiment are described.

When rotor 103 of motor element 104 rotates crankshaft 110, the rotation movement of eccentric shaft 112 is transmitted to piston 140 via connecting rod 146 and piston pin 142 as a connecting portion, and thereby piston 140 reciprocates in cylinder 131. When piston 140 reciprocates, a refrigerant gas is sucked from a cooling system (not shown) into cylinder 131, compressed and then discharged into the cooling system, again.

Next, an operation of oil supplying structure 120 is described. By the rotation of crankshaft 110, centrifugal pump 122 is rotated so as to generate centrifugal force. By the centrifugal force, oil 106 moves upwardly in centrifugal pump 122 to reach viscosity pump 121. Oil 106 which reached viscosity pump 121 further moves upwardly in viscosity pump 121 and is scattered in housing 101 via vertical hole 123 and lateral hole 124.

Oil 106 scattered in housing 101 collides with collision-portion 134 and moves along notch 135 so as to be attached to outer circumferential surface 150 of the piston. Attached oil 106 moves around outer circumferential surface 150, under cut 153, annular groove 191 and minute tapers 201 and 202 in accordance with the reciprocating movement of piston 140, and works as a lubricant between outer circumferential surface 150 and cylinder 131.

In the hermetic compressor of this exemplary embodiment, as shown in Fig. 1, in the vicinity of the bottom dead center, a part of the skirt side of piston 140 is protruded from cylinder 131. Therefore, when piston 140 comes to the bottom dead center, at least a part of under cut 153 is protruded from cylinder 131 and can be brought into direct contact with oil 106 scattered in housing 101. Thus, a sufficient amount of oil 106 is always supplied to under cut 153.

As shown in Fig. 5, oil 106 entering under cut 153 is accumulated in the

vicinity of edge 180 of under cut 153. When piston 140 moves from the bottom dead center to the top dead center, oil 106 is carried to an inner part of cylinder 131. On the other hand, when piston 140 moves from the top dead center to the bottom top dead center, in accordance with the movement of piston 140, oil 106 is drawn into between cylinder 131 and outer circumferential surface 150 of the piston so as to efficiently lubricate the vicinity of circumferentially formed land 190.

Furthermore, since angle θ made by edge 180 and outer circumferential surface 150 of the piton is made to be an acute angle, in accordance with the movement of piston 140, oil 106 is efficiently drawn into between cylinder 131 and outer circumferential surface 150 of the piston.

In this exemplary embodiment, since four under cuts 153 are provided in the axial direction of piston 140, through under cut 153, oil 106 is supplied to the wide range of outer circumferential surface 150 of the piston.

With the synergistic effect of them, the lubricant property of piston 140 is improved, and an extremely good sealing property can be obtained so as to suppress leakage of refrigerant. Therefore, high efficiency can be realized.

In general, when piston 140 is in the vicinity of the top dead center, the inside of cylinder 131 becomes high pressure due to a compressed refrigerant, so that a refrigerant gas is about to leak from between cylinder 131 and outer circumferential surface 150 of the piston. At this time, by compression load generated inside cylinder 131, via piston pin 142 and connecting rod 146, crankshaft 110 is pressed a direction opposite to the piston and may be inclined. When crankshaft 110 is inclined, piston 140 may be inclined in the vertical direction with respect to cylinder 131, thereby forming a part in which space between cylinder 131 and outer circumferential surface 150 of the piston may be broadened. As a result, leakage of a refrigerant gas from the part may be

accelerated. Furthermore, the inclination of piston 140 may deteriorate the lubricant state between piston 140 and cylinder 131 and may increase sliding noise.

However, in this exemplary embodiment, since upper sliding surface 154 and lower sliding surface 155 of piston 140 are provided over the full length of piston 140 from top surface 151 to skirt surface 152 as shown in Figs. 3 and 4, the inclination in the vertical direction of piston 140 is regulated, and thus the generation of inclination of piston 140 can be effectively suppressed. As a result of suppression of the inclination, leakage of refrigerant gas from cylinder 131 to housing 101 is suppressed, the behavior of piston 140 becomes stable, and it is possible to reduce sliding loss and to suppress the increase in noise. Consequently, high efficiency and low noise can be achieved.

Furthermore, the sliding loss generated when piston 140 reciprocates in cylinder 131 is in a state of fluid lubricant in which the loss is reduced in proportion to reduction of the sliding area. In this exemplary embodiment, since the area of under cut 153 is set to not less than one half of the area of outer circumferential surface 150 of the piston, sliding loss of piston 140 is about one half. Thus, high efficiency by remarkable input reduction can be realized.

Furthermore, during the compression stroke, a high pressure gas inside cylinder 131 leaks out to under cut 153. However, since under cut 153 always communicates with space inside housing 101 at the skirt surface 152 side, leaked refrigerant gas is not accumulated in under cut 153. Therefore, jet noise is not generated when the under cut comes out from the cylinder and a high pressure gas is released into low pressure space inside housing 101 at once in the case of a piston having a structure in which an under cut does not communicates with space inside housing 101. Furthermore, a high pressure

gas accumulated in the under cut does not backflow into cylinder 131 to increase re-expansion loss during the suction stroke.

Note here that, in this exemplary embodiment, under cut 153 always communicates with skirt surface 152. However, another configuration mentioned below can provide the same effect because a high pressure gas is released into space inside housing 101. That is to say, without allowing under cut 153 to communicate with skirt surface 152, under cut 153 may be allowed to communicate with space inside housing 101 only in the vicinity of the bottom dead center, or under cut 153 may be allowed to communicate with piston pin hole 141.

Furthermore, when circumferentially formed land 190 is provided with annular groove 191 and oil 106 is allowed to be brought into direct contact with annular groove 191 in the vicinity of the bottom dead center in which piston 140 is protruded from cylinder 131, attached oil 106 is spread over the entire part of annular groove 191 by the capillary phenomenon. Thereafter, during the movement of piston 140 from the bottom dead center to the top dead center, when a refrigerant gas reaches annular groove 191 and is joined together with oil 106 in groove 191, great viscosity resistance acts on the refrigerant gas. Furthermore, joined oil 106 and the refrigerant gas are expanded and contracted repeatedly, so that the pressure is reduced, whereby a so-called labyrinth seal effect is generated and the sealing property with respect to the leakage of refrigerant from cylinder 131 is improved. From the effect mentioned above, oil supply to the circumferentially formed land is further promoted, the lubricant property can be further improved, and furthermore, high efficiency can be achieved.

Next, the role of minute tapers 201 and 202 provided at the end portions both at the top surface 151 side and the skirt surface 152 side of piston

140 is described. When the piston moves from the bottom dead center to the top dead center, by the wedge effect of minute taper 201 at top surface 151 side of piston 140, oil 106 moves around circumferentially formed land 190 of piston 140 so as to improve the lubricant property of piston 140 and to also improve the sealing property. On the other hand, when piston 140 moves from the top dead center to the bottom dead center, by the wedge effect of minute taper 202 at the skirt surface 152 side, oil 106 enters minute taper 202 so as to form an oil film and lubricant property and sealing property are improved. That is to say, the presence of minute tapers 201 and 202 suppresses the leakage of refrigerant and reduces the sliding loss. Furthermore, high efficiency can be achieved.

Furthermore, in the case where the motor element is inverter-driven at plural operation frequencies including operation frequency that is not more than power supply frequency, reciprocating movement speed of piston 140 is reduced during low speed operation. Furthermore, since an amount of oil 106 scattered in housing 101 is reduced, leakage of refrigerant from space between outer circumferential surface 150 of the piston and cylinder 131 is likely to be increased. On the other hand, in the hermetic compressor of this exemplary embodiment, since oil 106 can be accumulated in under cut 153 and inclination in the vertical direction of piston 140 can be suppressed, high efficiency can be maintained also during the low speed operation.

The density of refrigerant R600a used in the hermetic compressor of this exemplary embodiment is smaller than the density of refrigerant R134a (1,1,1,2-tetrafluoroethane), which has been conventionally used in refrigerators. Therefore, when refrigerating ability that is the same as in a hermetic compressor using refrigerant R134a is intended to be obtained by using refrigerant R600a, cylinder capacity is increased and the outer diameter of piston 140 may be increased. Necessarily, the flow passage area for a

refrigerant is increased and the amount of refrigerant leaking into housing 101 from cylinder 131 is likely to increase. However, in the hermetic compressor of this exemplary embodiment, since the inclination of piston 140 with respect to cylinder 131 can be suppressed, the efficiency can be improved.

Note here that crankshaft 110 may be provided with a secondary axis which is provided on the same axis as main shaft 111 and opposed to main shaft with eccentric shaft 112 therebetween, and at the same time, a secondary bearing for supporting the secondary axis may be provided. With such a configuration, since crankshaft 110 is supported at both ends with eccentric shaft 112 sandwiched therebetween, the inclination of piston 140 in the vertical direction with respect to cylinder 131 is effectively suppressed. Consequently, since the behavior of piston 140 becomes stable, sliding loss can be reduced and the increase in noise can be suppressed, it is possible to realize a hermetic compressor with high efficiency and low noise property.

INDUSTRIAL APPLICABILITY

As mentioned above, since a hermetic compressor according to the present invention yields high productivity, and can increase efficiency and reliability, it can be widely applied to a hermetic compressor of, for example, an air conditioner, a vending machine, or the like.